NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED
March 1942 as
Advance Restricted Report

AN EXPERIMENTAL INVESTIGATION OF FLOW ACROSS TUBE BANKS

By M. J. Brevoort and A. N. Tifford

Langley Memorial Aeronautical Laboratory
Langley Field, Va.

NACA

WASHINGTON

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE RESTRICTED REPORT

AN EXPERIMENTAL INVESTIGATION OF FLOW ACROSS TUBE BANKS

By M. J. Brevoort and A. N. Tifford

L-232

SUMMARY

An experimental investigation of the details of the flow of fluids across tube banks has been conducted. Information that clarifies the picture of the flow has been obtained by surveys of total, dynamic, and static pressure, by visualization of the flow through the use of titanium tetrachloride smoke, by thermocouple surveys of heated tubes, and by hot-wire surveys of both heated and unheated tubes.

INTRODUCTION

The object of good design in heat exchangers is to obtain a high ratio of heat transfer to pressure drop. In the design of tubular intercoolers for aircraft, tubes of circular cross section are generally used. In reference 1 pressure-drop and heat-transfer data for various bank arrangements of such tubes have been published. At best, the ratio of heat transfer to pressure drop when the air passes across the tubes is half that obtained when air passes through the tubes.

The question then arises as to why flow across tube banks should be so much less efficient than flow through tubes. The answer is known in a general way. Flow through tubes is entirely frictional; whereas, in flow across tubes, the boundary layer breaks away from the rear surface of the tubes and vortices are shed. The shedding of vortices accounts for the low ratio of heat transfer to pressure drop.

The present work is a detailed study of the flow conditions existing in a bank of staggered circular tubes. The object of the study has been to clarify the physical picture of the phenomena occurring in tube banks.

SYMBOLS

C_D total drag coefficient (D/qF)

C_{D_o} form drag coefficient

D drag

D_h hydraulic diameter of passage

D_t diameter of tube

F frontal area

I current to Wheatstone bridge

I_o current to Wheatstone bridge when no air flows over hot wire

Δp pressure drop

q dynamic pressure $(\frac{1}{2}\rho V^2)$

s spacing

T_w temperature of tube wall

T_{ia} temperature of unheated air

V average velocity of air at minimum section between tubes in bank

ρ mass density of air

μ absolute viscosity of air

APPARATUS AND METHODS

The measurements presented in this report were taken in the duct shown in figure 1. The static pressure ahead of the bank of tubes was measured at the top wall at a distance of $1\frac{1}{2}$ feet downstream from the entrance. The static pressure was also measured at the top wall 2 feet behind the bank of tubes.

A tube bank was made of wooden circular tubes 8 inches long and 2.25 inches in diameter spaced transversely 2.49 inches and staggered longitudinally in such a way that a minimum of expansion and contraction was experienced as the air passed through the bank (fig. 2). The bank was composed of nine rows with four tubes in each row. Half tubes were used at the walls to fill out the tube bank and to reduce the effect of the side walls.

One tube near the middle of each row had surface orifices 180° apart for measuring pressures normal to the tube surface. (See fig. 3.) The tubes were set in the bank in such a position that the open ends of the tubing were 90° from the stagnation point of the tubes. Pressure-drop data in the bank were obtained by means of the static-pressure measurements of these tubes. One of the tubes was mounted on a pin and the entire tube was rotated to obtain data for the static-pressure distribution around the circumference of the tube. The effect of bank depth was obtained by adding rows to and subtracting rows from the bank.

Determinations of the flow near the surface of a tube were made with a hot wire mounted 0.014 inch from the surface of the rotatable tube. The hot wire was one arm of a Wheatstone bridge; the other three arms were fixed constantan resistances. The current supplied to the bridge was varied during the measurements to keep the hot wire at a constant temperature - just below red heat. The bridge had originally been so adjusted that the resistance of the wire corresponding to this temperature balanced the bridge. The amount of current required to balance the bridge at each position of the hot wire is a measure of the local velocity of the air. The hot wire itself was a 1-inch length of 0.003-inch-diameter platinum wire.

A brass tube of the same dimensions as the wooden tube previously described and containing an internal 350-watt heating coil was inserted into the bank. Hot-wire surveys of the heated surface were taken. In addition, an iron-constantan thermocouple was inbedded about 1/16 inch below the surface and a temperature survey of the tube surface was taken.

Static-pressure and total-pressure surveys were taken in the fifth row of the bank by means of the apparatus shown in figure 4. A 0.004-inch-diameter hole was drilled

in the side of the 0.030-inch-diameter tubing, which was closed at the lower end. The tube shaft extended about 4 inches below the top of the bank andcould be moved along a slit in the top of the duct. As the tube could be rotated, the drilled hole could be placed in any desired position around the axis of rotation.

RESULTS AND DISCUSSION

A survey of the total pressure, the static pressure, and the dynamic pressure 12 inches behind the bank of circular tubes was taken with a pitot-static tube. The traverse (fig. 5) showed that the static pressure was uniform across the duct cross section but that the total pressure varied according to the geometrical arrangement of the last row of the bank of tubes. The dynamic-pressure traverse shows these variations in greater detail. The straight line at 0.41 inch of alcohol, representing the dynamic pressure measured ahead of the bank, is lower than the average dynamic pressure measured behind the bank. The reason for this discrepancy is the large-scale turbulence, set up by the bank, causing the dynamic-pressure values behind the bank to be too high, as explained in reference 2. The variation of the dynamic pressure across the duct follows the geometry of the last row of tubes; that is, the dynamic pressure is high in the open spaces and low behind the tubes. The distributions of total, static, and dynamic pressure were found to be symmetrical about the center line and only half a section is therefore shown in figure 5.

VISUALIZATION OF THE FLOW THROUGH THE BANK

The front stagnation point of a tube in the first and fifth rows of the bank was painted with titanium tetrachloride and the path followed by the resultant smoke was observed through a transparent celluloid window. At very low Reynolds numbers the flow around the tube in the first row was similar to the potential-flow pattern shown in (a) of figure 6. There was no dead-air region at the rear of the tube and the wake formed a straight line filament extending to the next tube in the third row. When the airspeed was increased a little, the straight line filament was observed to oscillate slightly; further increases in

airspeed increased the magnitude of these oscillations until a complete von Kármán vortex street (fig. 6(b)) was formed. At a Reynolds number of 1000, a von Kármán vortex street existed behind the tube in the first row, whereas an unorganized shedding of vortices (fig. 6(c)) was observed behind the tube in the fifth row. The formation of a von Kármán vortex street behind the tube in the first row is analogous to the formation of a vortex street behind a circular cylinder in the free air stream (reference 3, pp. 227-228). The vortices behind the tube in the fifth row are a combination of the vortices shed by that tube and the vortices shed by the tubes in the rows farther forward. The superposition of these vortices accounts for their apparently unorganized appearance. As the airspeed was further increased, the vortex formation behind the first row appeared to become two symmetrical standing vortices (fig. 6(d)). At the same Reynolds number of 7000, the wake behind the tube in the fifth row was diffusedly turbulent (fig. 6(e)) and had no definite vortex structure. This difference in the flow in the first and fifth rows explains the entrance effect (reference 1). As the speed of the air increases, it takes a smaller number of rows of tubes to cause the flow pattern to become diffusedly turbulent.

TOTAL-PRESSURE AND DYNAMIC-PRESSURE SURVEYS IN THE BANK

A survey of the total and dynamic pressures in row 5 was taken by means of the apparatus shown in figure 4. At each point of the traverse the maximum reading - that is, the total pressure - was taken; the measuring hole was then rotated and a minimum reading taken. This minimum reading measures the total pressure minus a constant times the dynamic pressure. By the use of this relation, the variation in the dynamic pressure is obtained.

The marked variations of the total pressure and the dynamic pressure in a bank are shown in figure 7. Immediately behind a tube (survey I) there is a broad wake of low total-pressure fluid. As the fluid accelerates into the minimum free area between tubes, the low total-pressure fluid picks up energy (surveys II and III). By the time the next tube two rows down the bank is reached, the total pressure is almost uniform across the cross section (survey IV). The traverse of dynamic pressure directly behind a tube (survey I) shows a broad wake of the

low dynamic-pressure fluid. As the fluid accelerates into the minimum area between tubes (surveys II and III), however, the low-velocity fluid in the center does not accelerate any faster than the rest of the fluid. Consequently, even at the minimum area between tubes (survey III), the central fluid has a very low velocity. The velocity of the central fluid again decreases to zero (survey IV) as the stagnation point of the next tube downstream is approached.

These surveys make it clear that, when fluid flows across a bank of tubes, the effects of the wake behind individual tubes extend to the following tubes. The two principal phenomena occurring are the formation of a "dead-air" region behind each tube, which prevents the main fluid stream from expanding appreciably, and the maintenance of a low-velocity sheet down the center of the open space between tubes. This sheet of low-velocity fluid reacts upon the adjacent fluid as if it were a semisolid wall that, together with the neighboring tube walls, forms a frictional passage through which the main fluid stream flows. The hydraulic diameter that is used in describing the frictional characteristics of the flow is the hydraulic diameter of this "pseudo-passage," which has the same dimensions as half the minimum passage between tubes of the same row.

FORM DRAG COEFFICIENTS OF CIRCULAR TUBES

The pressure distributions at the tube surface in various rows of the bank investigated at two Reynolds numbers, 42,300 and 70,500, based on tube diameter, were similar to the pressure distributions presented in reference 4. These pressure distributions were used to obtain the form drag coefficients of the tubes. The drag coefficient of a tube in the bank is not comparable with the drag coefficient of a tube in the free air stream because of the difference in the velocity fields in the two cases. Figure 8 shows the form drag coefficient as a function of the bank depth. The variation of the form drag coefficient with the bank depth is easily explained by the smoke surveys described earlier. The flow over the first row of the bank is laminar ahead of breakaway just as it would be in the case of a tube in the free air stream. The flow over the second row differs from laminar flow by the amount of oscillation of the flow caused by the first row.

L-232

The flow over the third row is affected by the oscillations caused by the first and the second rows and so on with the succeeding rows. The relatively undisturbed flow over the first few rows causes their form drag coefficients to be high just as the form drag coefficients of circular cylinders in the free air stream are high when the flow is very laminar. After three or four rows have been passed, the flow becomes very turbulent and the form drag coefficients thereafter are constant. The last row has a much lower form drag than have the preceding rows because of the reduced dynamic pressure and the accompanying increased static pressure at the rear of the row.

TOTAL DRAG COEFFICIENTS OF CIRCULAR TUBES

Figure 9 shows the pressure-drop data obtained from a 9-row bank of circular tubes. The average pressure drop per row was added to the over-all pressure drop across the 9-row bank to obtain values for plotting the dashed line representing the pressure drop across a 10-row bank. Data from measurements on a 10-row bank fall along this line, which serves as a check upon the measurements.

The total drag coefficients of circular tubes in a bank are obtained from the pressure-drop data by means of the relation

$$C_D = \left(\frac{\Delta p}{q}\right)_{row} \frac{s + D_t}{D_t}$$

At a Reynolds number of 42,300 the total drag coefficient is 0.539. The form drag coefficient was measured earlier as 0.468 and the difference, which is the friction drag coefficient, is 0.071. Friction accounts for 13.2 percent of the total drag. Similarly at a Reynolds number of 70,500 the total drag coefficient is 0.495, the form drag coefficient is 0.427, and the friction drag coefficient is 0.068. Friction accounts for 13.7 percent of the total drag. In reference 5 the friction drag of a single circular tube in the free air stream was found to be 2 percent of the total drag at a Reynolds number of 40,000. The results show that the inclusion of the tube in a bank spaced with $s/D_t = 1.106$ increases the friction drag to 13 percent of the total drag.

The total drag coefficient at a Reynolds number of 20,000 as a function of the spacing of the tube bank has been plotted in figure 10 from faired data from reference 1. The data obtained agree well with this plot. The theoretical friction line is based upon the frictional hydraulic diameter discussed earlier. The two experimental points agree surprisingly well with the theoretical friction line considering that they are calculated as the difference between two large quantities, the total drag coefficient and the form drag coefficient.

HOT-WIRE SURVEYS OF CIRCULAR TUBES

Figure 11 shows typical data obtained in hot-wire surveys of the tube surface in rows 1, 2, and 4 and in the last row. The entrance effect is clearly seen in rows 1 and 2. There is early breakaway from the tube surface in these rows because of the relatively laminar air flow described earlier. The data for row 4 are typical of tubes deep in the bank. A peak in cooling occurs at between 70° and 90° from the front stagnation point. Tubes in the last row show a large region of breakaway, which is, of course, a result of the large expansion behind the bank.

Figure 12 gives the results of hot-wire surveys on a circular tube in the free air stream at three Reynolds numbers: 8,900, 23,700, and 41,500. These results indicate that the front of the tube is more effective in cooling than the rear of the tube.

Hot-wire surveys of an internally heated brass tube placed in various rows of the bank agree, in general, with the earlier surveys (fig. 11) but show progressively less cooling toward the rear of the tube (fig. 13) because of the increasing temperature of the air film. (Similar results were obtained for a single tube in the free air stream.) Figure 14 shows typical data obtained by means of a thermocouple survey of the heated tube surface at a Reynolds number of 42,300. The average temperature difference available for cooling was 3.37 millivolts, which corresponds to 128.4° F, and the amount of heat dissipated was 0.244 Btu per second. The average heat-transfer coefficient was therefore 0.00483 Btu per second per $^{\circ}$F per square foot. When the thermal conductivity of the air at the "film" temperature of $T_w - \frac{1}{2}(T_w - T_{ia})$ (reference 1)

REFERENCES

1. Pierson, Orville L.: Experimental Investigation of the Influence of Tube Arrangement on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases over Tube Banks. A.S.M.E. Trans., PRO-59-6, vol. 59, no. 7, Oct. 1937, pp. 563-572.

2. Goldstein, S.: Modern Developments in Fluid Dynamics. Vol. I. The Clarendon Press (Oxford), 1938, pp. 253-254.

3. Hollingdale, S. H.: Stability and Configuration of the Wakes Produced by Solid Bodies Moving through Fluids. Phil. Mag., ser. 7, vol. 29, no. 194, March 1940, pp. 211-257.

4. Wallis, R. Pendennis: The Optimum Size of Models for Studying Flow through Nests of Tubes. Engineering, vol. 147, no. 3824, April 28, 1939, pp. 487-489.

5. Schiller, L., and Linke, W.: Pressure and Frictional Resistance of a Cylinder at Reynolds Numbers 5,000 to 40,000. T.M. No. 715, NACA, 1933.

is used, Nusselt's number is calculated to be 204.5, which agrees with the data of reference 1.

L-232

CONCLUDING REMARKS

Several experimental methods have been used in this study and new information that clarifies the picture of the flow of fluid across tube banks has been obtained. Titanium-tetrachloride smoke surveys clearly showed the increase of the turbulence of the flow with the depth of the bank. Surveys of the static pressure at the surface of tubes in each row indicated that, at a Reynolds number of 40,000, the effects of the smoother flow in the first few rows extended at least three rows into the bank. At the slightly lower Reynolds numbers at which aircraft heat exchangers usually operate, smoke surveys indicated that the entrance effects extended about four rows into the bank.

Total-pressure and dynamic-pressure surveys deep in the bank revealed that a broad wake behind each tube affected the flow over the succeeding tubes. Because of the wake, a sheet of low-velocity fluid was maintained down the center of the space between tubes. The air flowed frictionally through the passage between this sheet and the tube surface. When the bank spacing was 1.11 times the tube diameter, friction accounted for 13 percent of the pressure drop. As the spacing decreased, the amount of friction increased.

Hot-wire surveys of the surface of tubes in each row showed that, for tubes deep in the bank, a peak in cooling occurred at between 70° and 90° from the front stagnation point of the tube and a slight peak occurred at the rear of the tube. A comparison of the hot-wire surveys of the surface of heated tubes with the hot-wire surveys of the surface of unheated tubes both in a bank and in the free air stream showed that the only effect of heating was progressively less cooling toward the rear of the tube. There was no evidence that the vortices at the rear of the tube were composed of cool air, as has sometimes been maintained.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va.

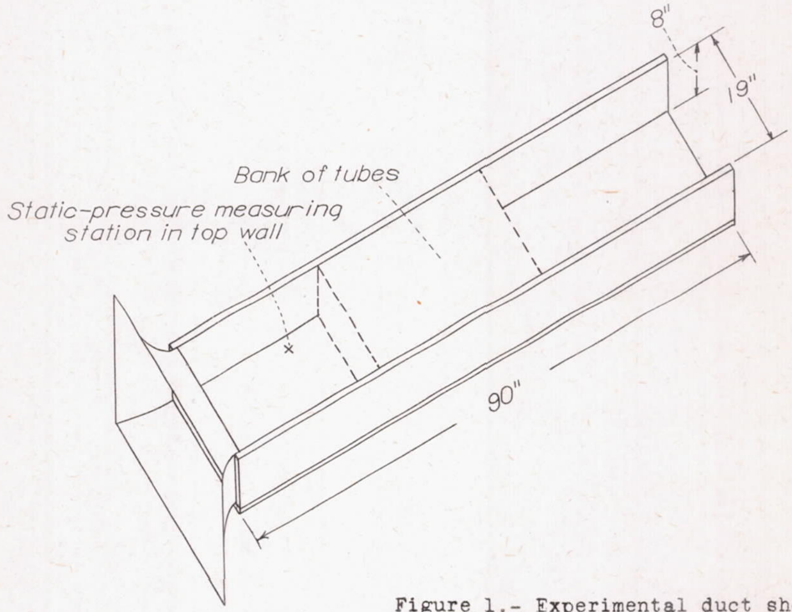


Figure 1.- Experimental duct showing position of bank of tubes.

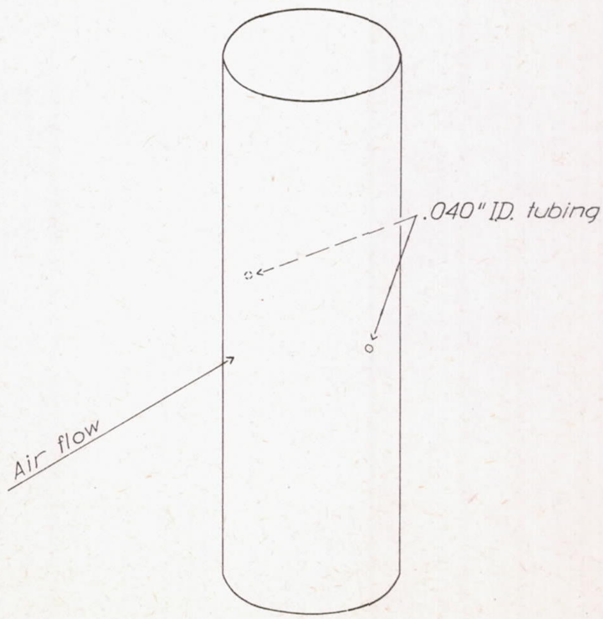


Figure 3.- Tube with static-pressure measuring tubing.

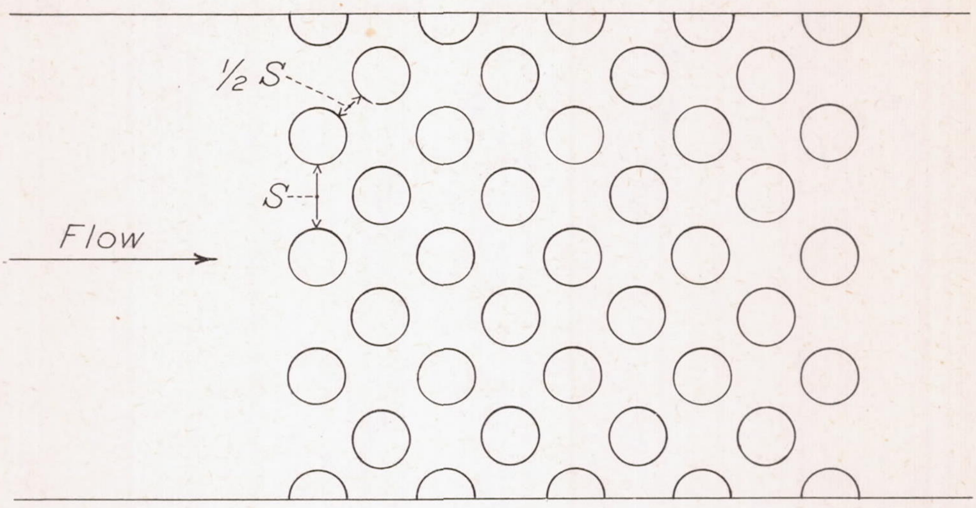


Figure 2.- Cross section of tube bank showing arrangement and spacing of tubes.

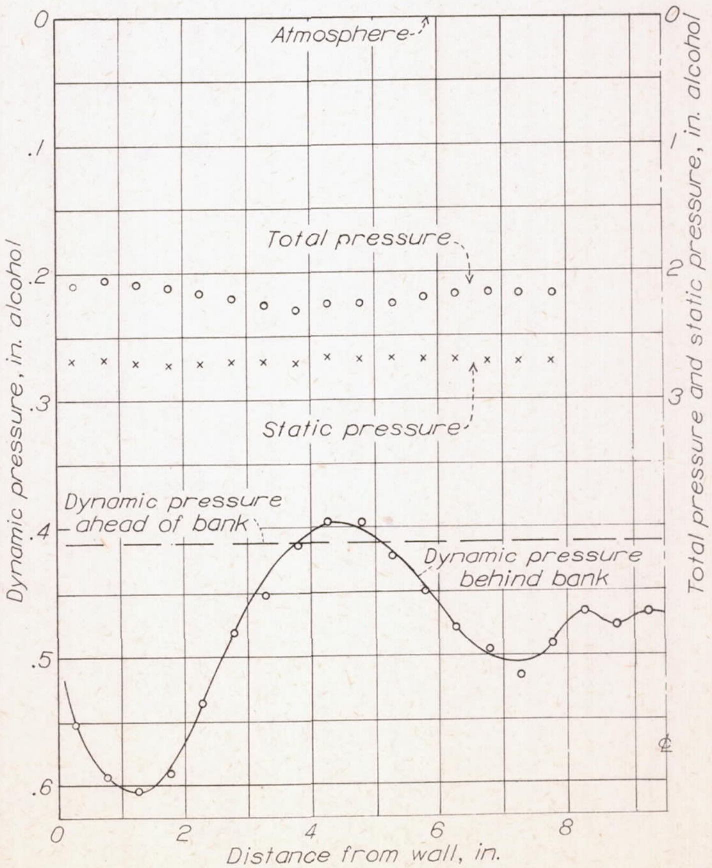


Figure 5.- Survey 12 inches behind bank of circular tubes.

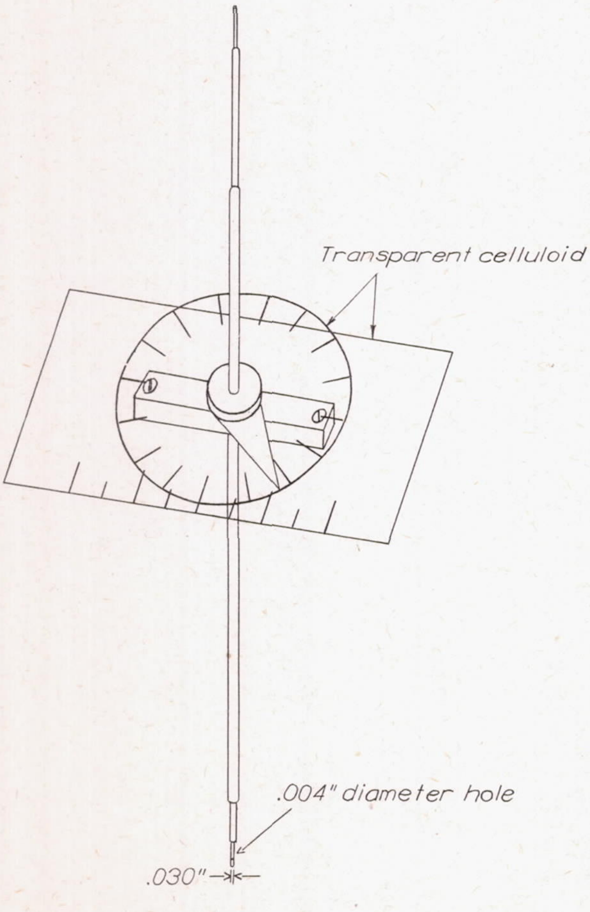


Figure 4.- Apparatus for pressure surveys inside bank.

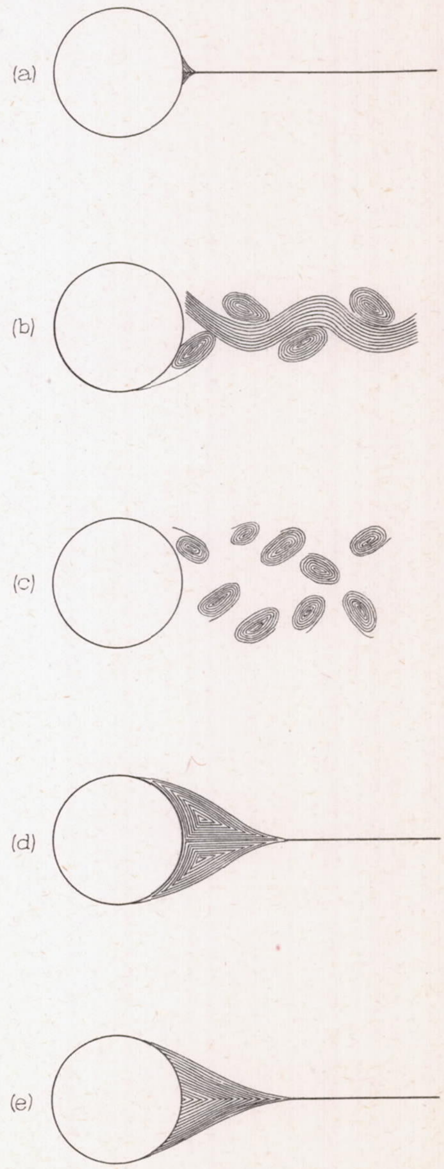


Figure 6.- Flow patterns.

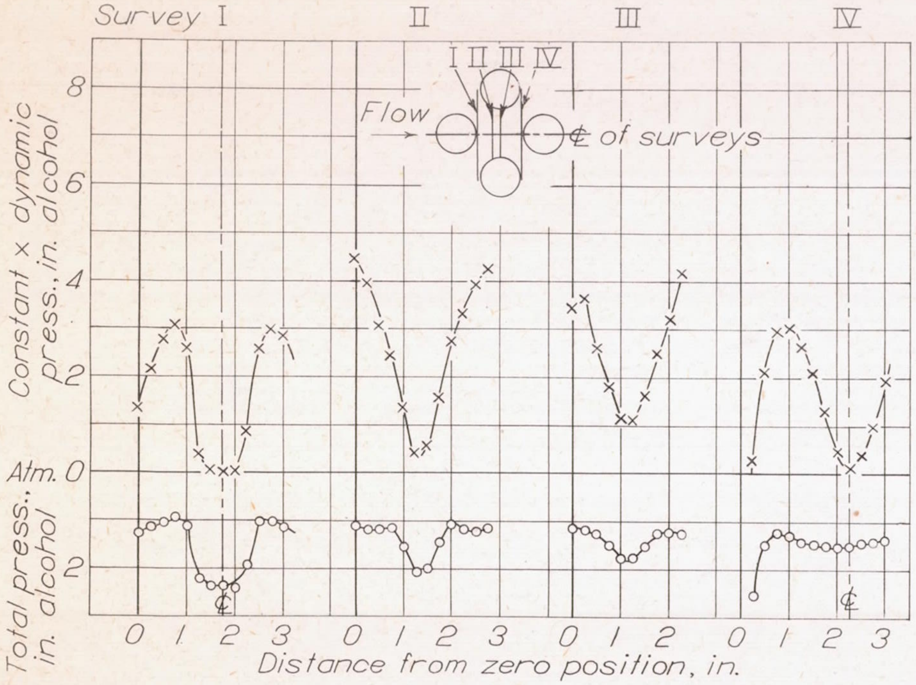
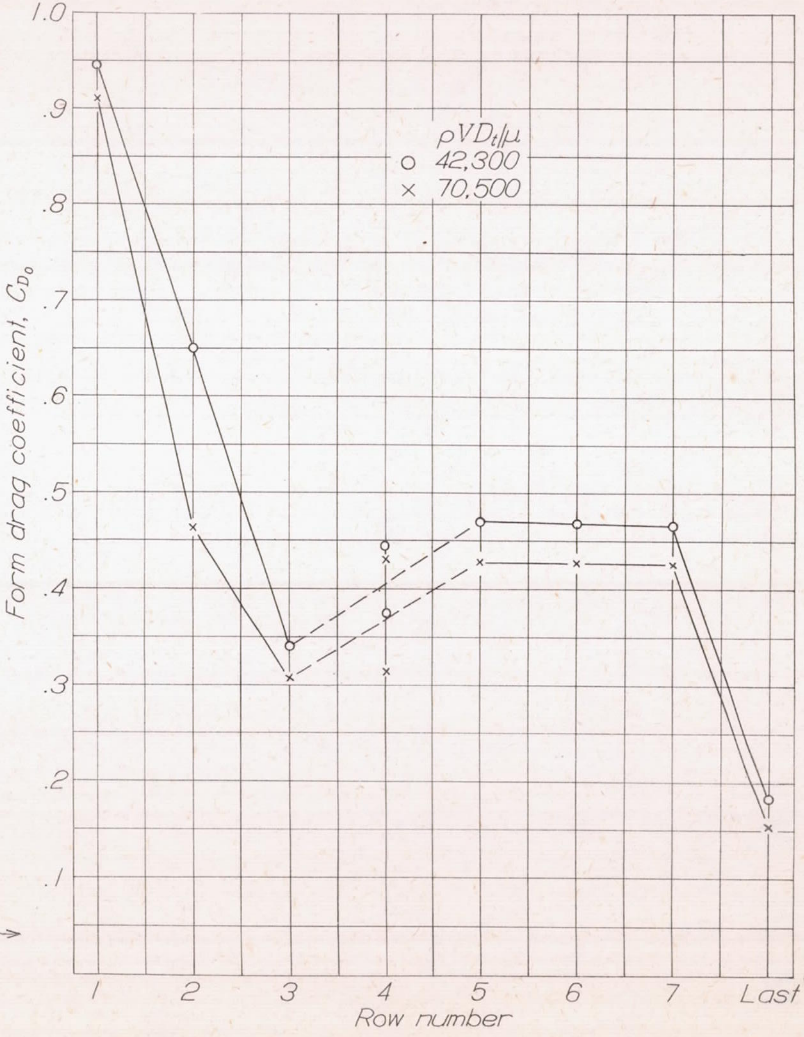


Figure 7.- Total-pressure and dynamic-pressure surveys in bank of circular tubes at $\rho VD_t/\mu = 82,000$.

Figure 8.- Form drag coefficient as a function of bank depth.



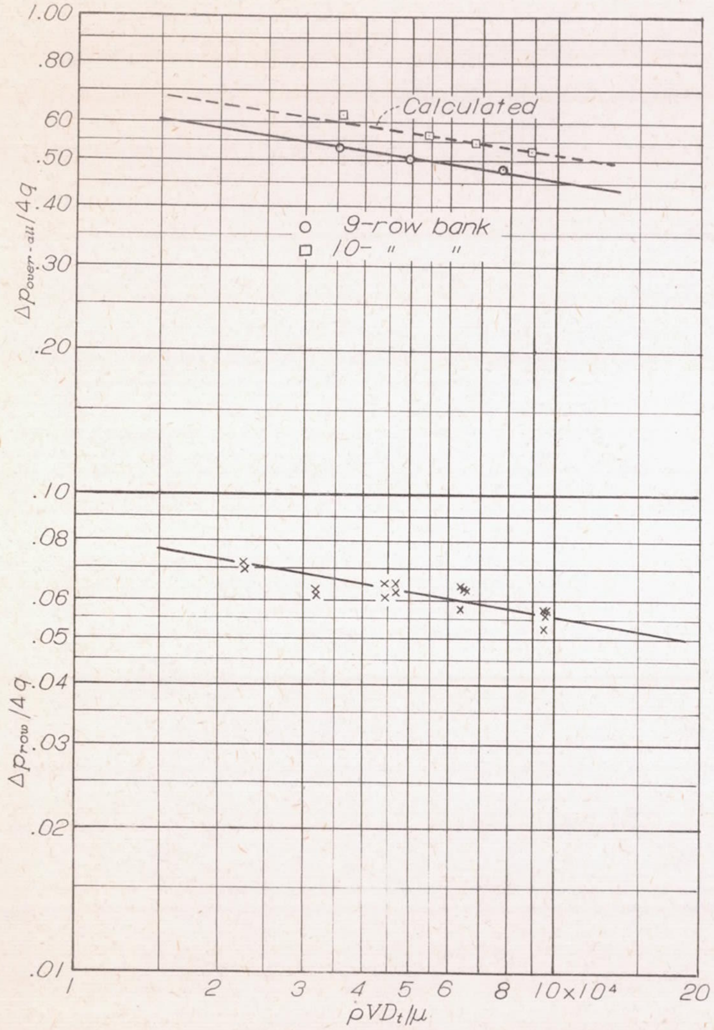


Figure 9.- Pressure-drop data for bank of circular tubes.

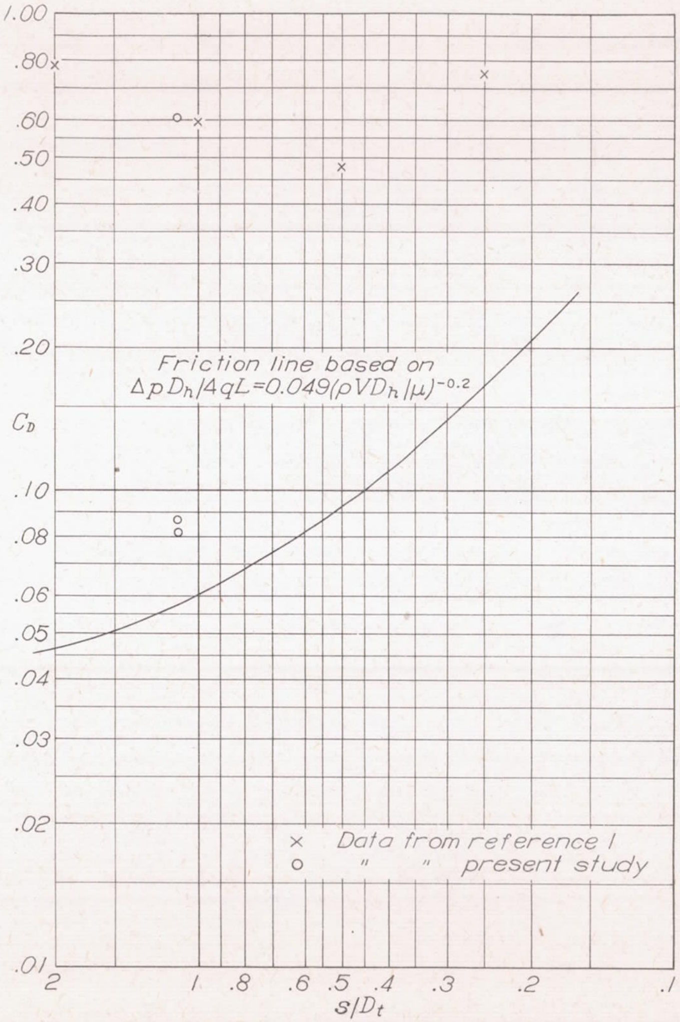


Figure 10.- Total drag coefficient as a function of spacing at $\rho V D_t/\mu$ = 20,000

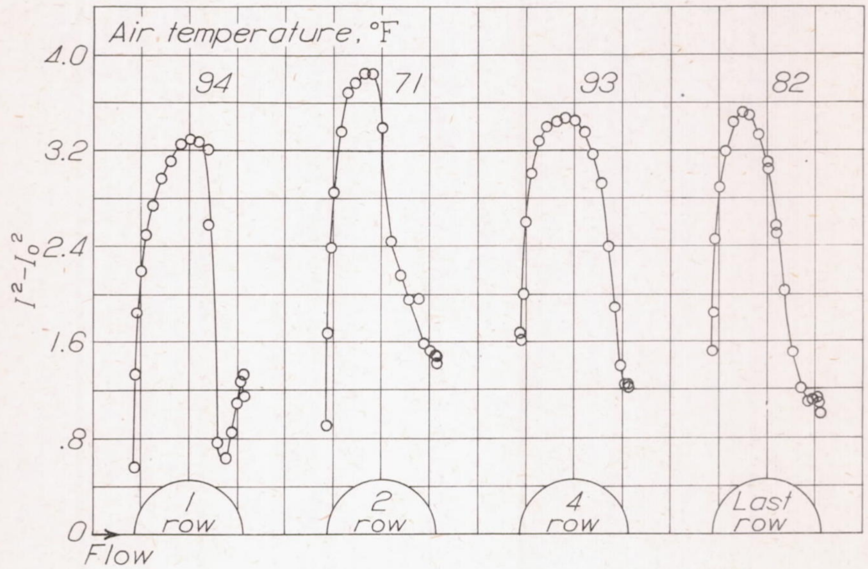


Figure 11.- Hot-wire surveys of surface of circular tubes at $\rho V D_t/\mu$ = 74,000.

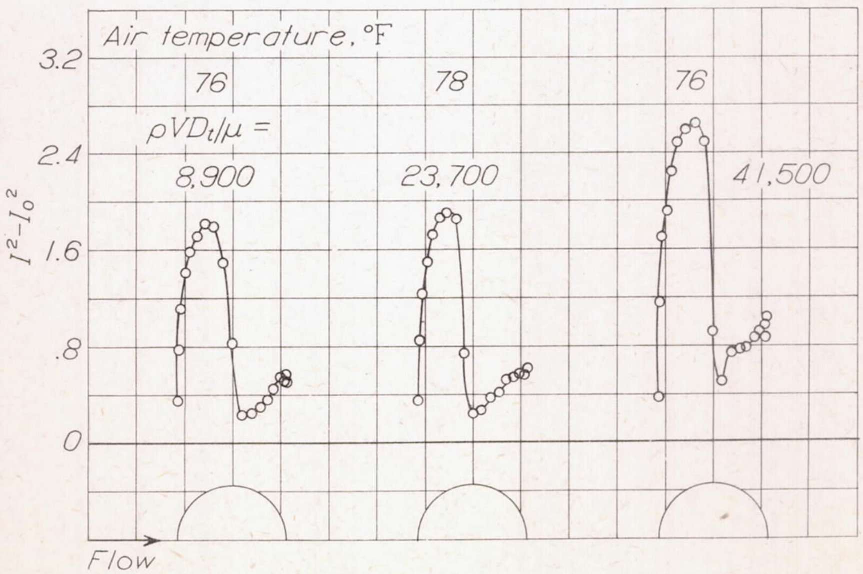


Figure 12.- Hot-wire surveys of circular cylinder in free air stream.

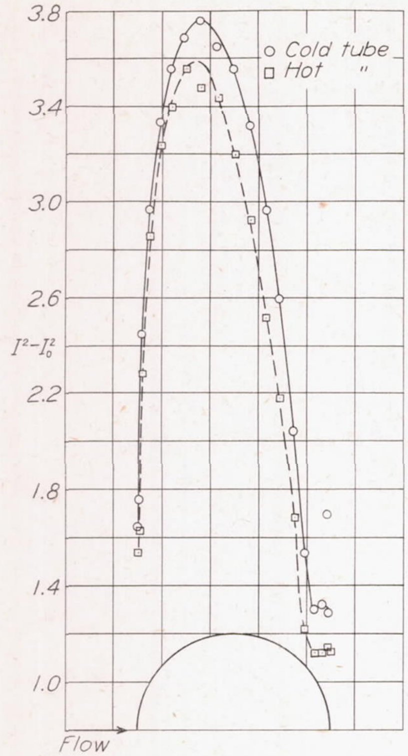


Figure 13.- Hot-wire surveys of the surface of the brass tube in row 3 at $\rho VD_t/\mu$ = 74,000. Air temperature = 67° F.

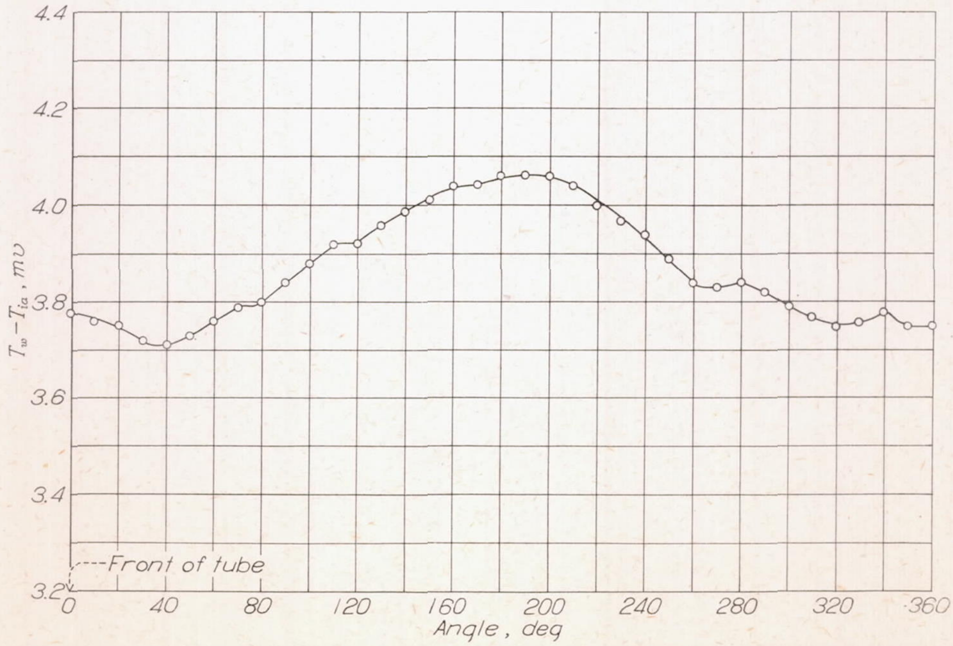


Figure 14.- Thermocouple survey of the surface of a heated brass circular tube in a bank at $\rho VD_t/\mu$ = 42,300. $(T_w-T_{ia})_{average}$ = 128.4°F, T_{ia} = 68.0°F.